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COLD-AIR INVESTIGATION OF A TURBINE WITH TRANSPIRATION-COOLED STATOR BLADES

I - Performance of Stator With Discrete Hole Blading

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16. Abstract

The effects of transpiration coolant ejection on stator mass flow and efficiency were experimentally determined for ratios of coolant to primary mass flow from 0 to 7 percent. The results showed that transpiration coolant discharge caused a decrease in total mass flow and efficiency. When compared with previously obtained experimental results for trailing-edge discharge, the performance loss for transpiration discharge was significantly larger. The predicted decrease in efficiency resulting from transpiration discharge was less than the experimental decrease.

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COLD-AIR INVESTIGATION OF A TURBINE WITH

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SUMMARY

At the NASA Lewis Research Center, the effects on turbine performance of different means of stator-blade coolant ejection are being investigated. This report presents the results of a detailed experimental and analytical investigation of the effects on stator performance of stator-blade transpiration coolant discharge. The blading investigated had discrete cooling holes in the outer surface. The investigation included the effects on stator mass flow and efficiency over a range of coolant mass flows from 0 to about 7 percent of primary mass flow.

Relative to the zero-coolant-flow results with the coolant holes sealed, the experimental results showed that the transpiration discharge of coolant caused a small decrease in total mass flow and primary air efficiency and a quite large decrease in thermodynamic efficiency. (The primary air efficiency relates the actual kinetic energy output to the ideal energy of the primary flow only; the thermodynamic efficiency charges the stator with the ideal energy of both the primary and coolant flows.) Over the range of coolant flows of practical interest, from about 3 to 7 percent, the decrease in total mass flow was a nearly constant value of 1.5 percent; the decrease in primary air efficiency was nearly constant at about 1.0 percent; and the decrease in thermodynamic efficiency was about 4.5 and 12.5 percent, respectively, at $3\frac{1}{4}$ and 7 percent coolant flow. When compared with corresponding efficiency results for stator blade trailing-edge coolant discharge, the penalty in efficiency for stator transpiration coolant discharge is large. The large penalty for transpiration coolant discharge relative to trailing-edge discharge is caused by the difference in location and direction of coolant discharge for the two methods.

A comparison was made of predicted and experimental effects of transpiration discharge on primary air efficiency over the coolant flow range between 3 and 7 percent. Relative to the efficiency with zero coolant and the holes sealed, the predicted results showed about a 1/4-percent gain in primary air efficiency. The experimental results showed about a 1-percent loss in primary air efficiency.

INTRODUCTION

To meet their performance requirements, many advance gas turbine engines must operate at high turbine inlet temperatures. In some cases, these temperatures are high enough that the turbine blading must be cooled to avoid exceeding the stress and oxidation limitations of currently available materials. The general method for cooling the blading is to bleed air from the compressor, direct it through the turbine blading for cooling, and then discharge it from the blading into the main gas stream. The means of discharging the coolant from the blading into the main gas stream are different for different cooled blade designs. In reference 1, an analysis is presented which indicates that different means of coolant ejection affect turbine performance quite differently. To confirm these findings experimentally, an investigation is being conducted at the NASA Lewis Research Center to determine the effect on turbine blade row and stage performance of several means of coolant discharge that are typical of different cooled blade designs.

The first coolant discharge method investigated was stator-blade trailing-edge ejection. In that investigation, the coolant was discharged into the main gas stream through a slot along the entire length of the stator-blade trailing edge. The results of the investigation, which include the effects of stator-blade trailing-edge ejection on both stator and stage performance, are reported in references 2 to 4. The results show that the flow of coolant caused only small changes in both stator and stage thermodynamic efficiency. From the stator and stage results, it was also deduced that the rotor efficiency was not significantly affected by the flow of coolant.

An investigation of the effect of stator-blade transpiration coolant ejection on turbine stator and stage performance is now being made. The stator investigation of the initial type of transpiration blading is complete. (Except for trailing-edge geometry, the profiles and primary flow path of this blading are the same, within manufacturing tolerance, as those used for the investigation of trailing-edge discharge.) With transpiration cooling, the coolant is introduced into hollow blades and then ejected through a porous shell that constitutes the entire outer blade surface. The particular blading tested has discrete cooling holes with variable porosity over the blade surface and a self-supporting shell. The self-supporting shell is somewhat unusual in that it eliminates the need for support struts with separate chambers and metering orifices associated with the more common wire-mesh shell-type transpiration-cooled blading.

This report presents the results of a detailed investigation of the performance of the described stator with self-supporting, variable-porosity, transpiration-type blading. The results include the effects of transpiration coolant discharge on stator mass flow characteristics, outlet flow angle, pressure loss patterns downstream of the blading, specific output, and overall efficiency. Selected results for the subject stator are also compared with corresponding results for the stator having trailing-edge ejection of

coolant flow. In addition, the experimental effects of transpiration coolant discharge on stator efficiency are compared with predicted effects.

During this investigation, the inlet temperature of the primary and coolant flow was the same, so the results include no actual coolant effects. The results are for nominal coolant flow rates of 0, 3, 5, and 7 percent of the primary mass flow. Mass flow and outlet flow angle results were obtained for a range of stator-inlet to downstream pressure ratios, while stator output and efficiency results were obtained at a constant stator inlet-to-downstream hub pressure ratio of 1.72, corresponding to stator design conditions (ref. 5).

SYMBOLS

Δh specific kinetic energy output, Btu/lb; J/kg

Δh_{pr} stator kinetic energy output per unit of primary air mass flow, Btu/lb; J/kg

Δh_{tot} stator kinetic energy output per unit of total (primary plus coolant) mass flow, Btu/lb; J/kg

p pressure, psi; N/m^2

V absolute gas velocity, ft/sec; m/sec

w mass flow rate, lb/sec; kg/sec

y coolant fraction, ratio of coolant to primary mass flow

α flow angle at blade outlet measured from axial direction, deg

δ ratio of stator inlet pressure to U.S. standard sea-level pressure

δ^* displacement thickness parameter; loss in flow as fractional part of ideal flow

η stator efficiency

η_p stator primary air efficiency; stator kinetic energy output expressed as fraction of ideal energy of primary flow only

η_t stator thermodynamic efficiency; stator kinetic energy output expressed as fraction of ideal energies of both primary and coolant flows

θ_{cr} squared ratio of critical velocity at stator inlet to critical velocity of U.S. standard sea-level air

Subscripts:

ave   average of hub and tip conditions

c   coolant flow

cl   blade coolant holes closed

cr   conditions at Mach 1

d   measuring station downstream of blading

h   blade hub

i   ideal quantity corresponding to isentropic process

o   absence of coolant flow

op   blade coolant holes open

p   primary flow

tot   total flow (primary plus coolant flow)

0   measuring station upstream of blading

1   measuring station at stator throat

2   measuring station just downstream of stator trailing edge

3   station where complete mixing occurs

3d   three dimensional or annular sector

Superscript:

′   total state

APPARATUS

The apparatus used in the investigation consisted of the test stator and an appropriate test facility. The facility, which is shown in the sketch of figure 1 and the photograph of figure 2, is the same as that described in reference 3.

The stator blading of the subject investigation is shown in figure 3. The blading is hollow cored with self-supporting porous walls. The outer surface of the blading has a large number of discrete small circular holes. The flow path of the coolant is from the hollow core of the blading, through the permeable blade walls, and out these holes at varying rates over different parts of the blade surface. Figure 4 shows the mean-section profile and flow path of the blading. The profiles and primary air flow path of this blading are the same, within manufacturing tolerance, as those of the stator blading described in detail in reference 5 and also the same, except for trailing-edge geometry, as those of the stator blading used to investigate stator-blade trailing-edge ejection of coolant flow (refs. 2 and 3). The following are significant dimensions of the blading: 30-inch (76.2-
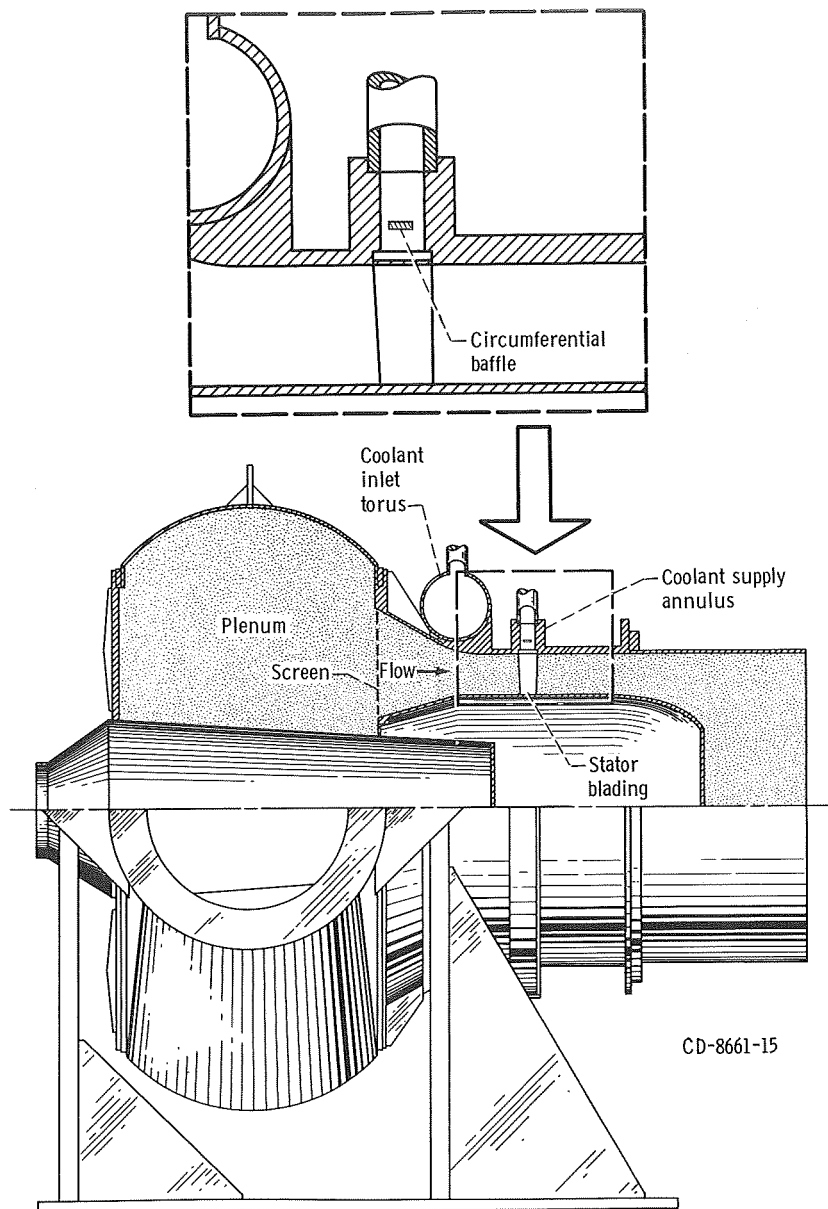
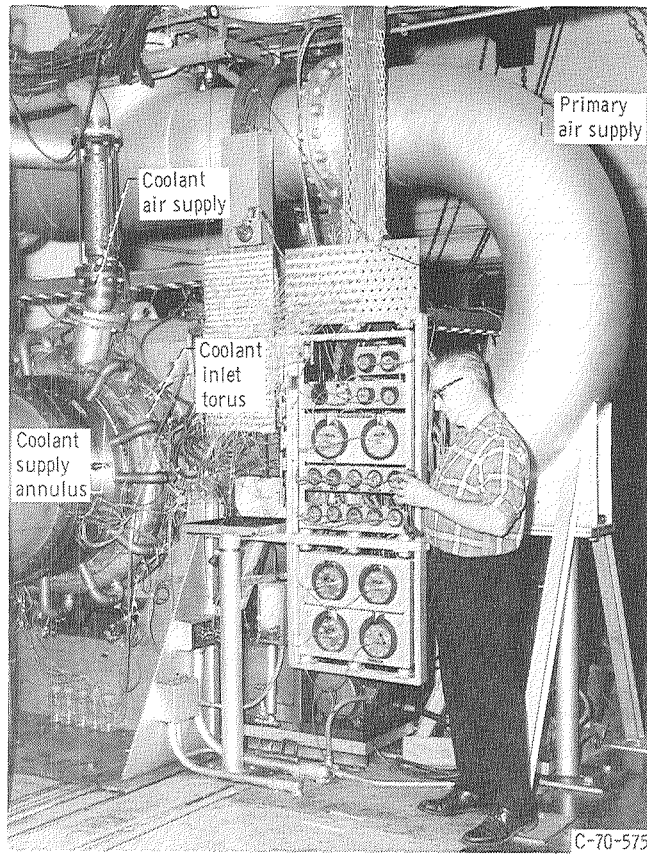Figure 1. - Cross section of test facility.

Labels in figure:
- Circumferential baffle
- Coolant inlet torus
- Coolant supply annulus
- Plenum
- Screen
- Flow
- Stator blading
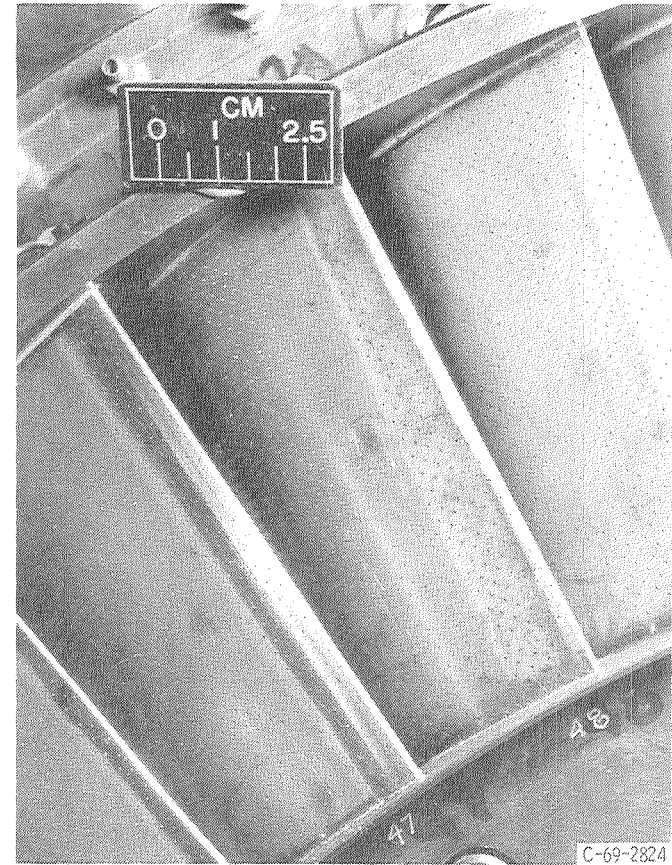- CD-8661-15

Figure 2. – Test facility.
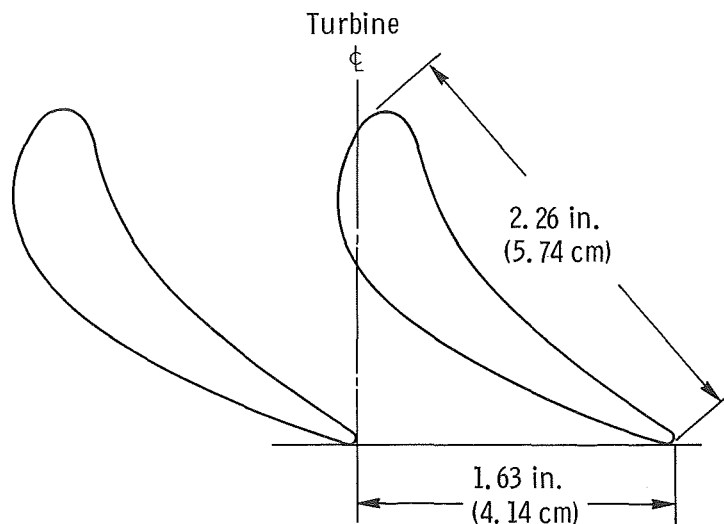

Figure 3. – Stator blading.

Figure 4. - Blade mean-section profile and flow path.

cm) tip diameter, 4.0-inch (10.16-cm) radial height, 2.26-inch- (5.74-cm-) mean-radius chord, and 1.63-inch- (4.14-cm-) mean-radius pitch.

INSTRUMENTATION

The instrumentation required for obtaining the reported results is the same as that described in reference 3. Briefly, a Dall-tube flowmeter was used to measure the primary flow rate, and a Venturi meter was used to measure the coolant flow rate. In the stator test section, total and static pressures were measured at the locations shown in figures 5 and 6. Inlet total pressures were measured with Kiel-type total pressure probes; downstream total pressures were measured with a total pressure survey probe of the modified type described in detail in reference 6; and static pressures were measured using conventional static pressure taps.
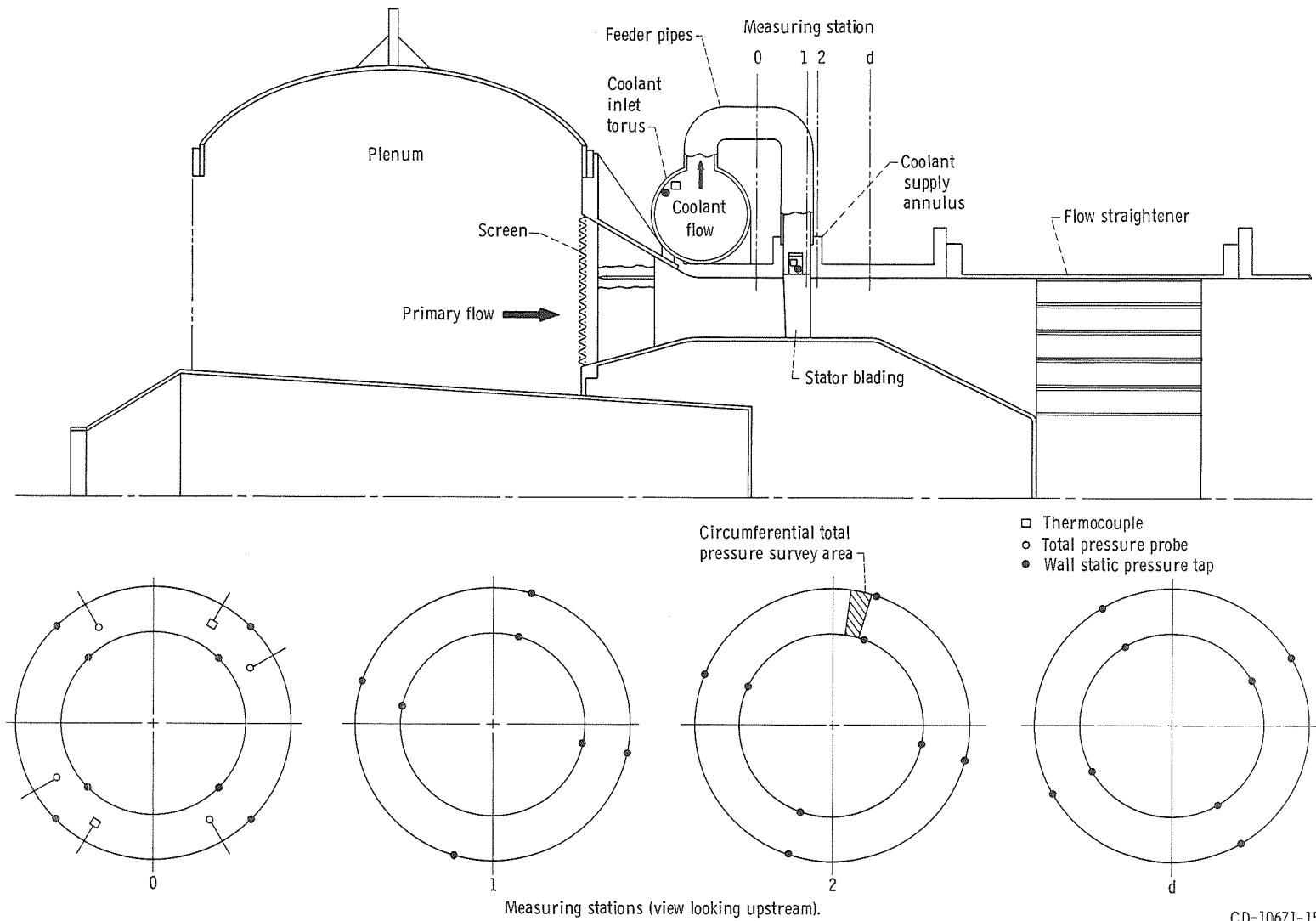
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Feeder pipes

Measuring station

0   1 2   d

Coolant
inlet
torus

Plenum

Coolant
flow

Coolant
supply
annulus

Flow straightener

Screen

Primary flow

Stator blading

Circumferential total
pressure survey area

□ Thermocouple
o Total pressure probe
● Wall static pressure tap

0

1

2

d

Measuring stations (view looking upstream).

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Figure 5. - Turbine stator test facility and location of instrumentation.
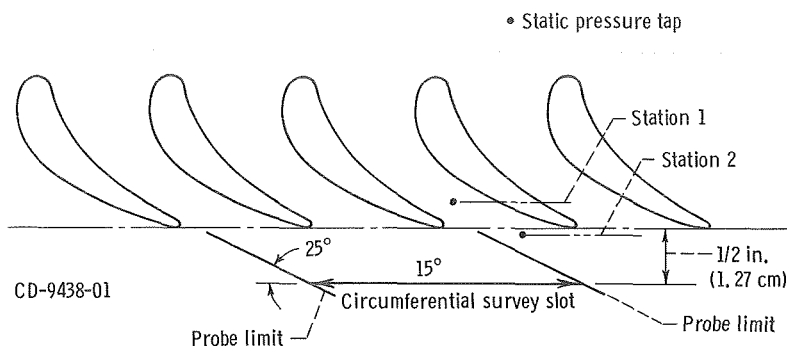
Figure 6. - Circumferential total pressure survey area with approximate location of wall static pressure taps.

TEST AND CALCULATION PROCEDURES

The fluid used in the investigation for both the primary and coolant flow was laboratory combustion air at a total temperature of about 545° F (303 K). The stator-inlet pressure of the primary air was 30 inches of mercury absolute (1.0159×10^5 N/m^2 abs).

Mass flow characteristics and outlet flow angle results were obtained for a desired range of overall stator pressure ratios $p_0'/p_{d,h}$ and coolant flow rates. Pressure ratios were set by holding the upstream pressure constant and adjusting the downstream pressure. At each pressure ratio setting, the coolant mass flow rate was varied by regulating the inlet pressure of the coolant. The stator-outlet flow angles were measured by traversing radially across the blade span with the probe situated midway between wakes. When the flow angles were measured, the probe was located about 0.5 inch (1.27 cm) axially downstream of the stator-blade trailing edge.

Stator output and efficiency results were obtained at a constant overall stator hub pressure ratio of 1.72, corresponding to design conditions for the stator of reference 5, with nominal coolant flow rates of 0, 3, 5, and 7 percent of the primary flow. At each coolant flow rate, annular surveys of total pressure loss were made for approximately one blade pitch (fig. 6). These annular surveys were made by conducting circumferential surveys at a sufficient number of different radii to define adequately the loss for the complete annular sector.

At zero-coolant flow, two annular surveys were conducted. For one of these surveys, the blade coolant holes were filled and smoothed flush to the outer blade surface. The purpose of this survey was to determine the performance of the stator without coolant provisions. In the other survey with zero-coolant flow, the blading was unaltered; that is, the coolant holes were left open. Under these conditions, some of the primary air flows through the interior of the blading. This condition of operation is unrealistic for transpiration-cooled blading, so the results would normally be only of academic

interest. However, for this investigation, the results were also of practical use, as now described.

To determine the stator mass flow with the coolant holes sealed using Dall-tube measurements would have required that all 50 blades of the stator be sealed, since the Dall-tube flow includes the flow to all blading. To seal all blades would have required a complete stator disassembly and reassembly. The stator mass flow with coolant holes sealed was therefore obtained by an indirect method that required that only three blades in the total pressure survey area be sealed. This modification was accomplished without disassembling the stator.

Briefly, the indirect procedure consisted of the following: mass flow results for zero-coolant flow with the coolant holes open were first determined from Dall-tube measurements. Then, the ratio of the actual mass flow to the ideal mass flow was determined for zero-coolant flow both with the coolant holes open and with the coolant holes sealed. These ratios were determined from annular-sector total pressure loss data using the procedures described in reference 7. Knowing the stator flow for zero coolant with the holes open and the ratio of actual to ideal stator flow for zero coolant with the holes both open and sealed made it possible to compute the stator mass flow with coolant holes sealed:

$$w_{o, cl} = w_{o, op} \left(\frac{1 - \delta^*_{o, 3d, cl}}{1 - \delta^*_{o, 3d, op}} \right)$$

where δ^*, the displacement thickness parameter, represents the fractional loss in ideal flow of the blade row (see ref. 7).

Except for the results with the coolant holes sealed, the test and calculation procedures used in determining the stator mass flow results were those conventionally used for Venturi- and Dall-tube-type flowmeters. Output and efficiency results were determined using the procedures summarized in reference 3.

The predicted effect of transpiration discharge on stator output was obtained by using the following assumptions and procedures:

(1) The coolant flow rate per unit of blade surface was constant.

(2) Since the velocity of coolant at the point of discharge on the blade surface is generally perpendicular to that of the main gas stream, the dynamic head of the coolant at the point of discharge is lost.

(3) The coolant-flow ideal available energy, corresponding to the static head between the point of discharge on the blade surface and blade exit conditions, is converted to useful kinetic energy at the same efficiency as the primary flow.

(4) The coolant flow and the primary flow mix downstream of the blade trailing edge.

(5) The efficiency of the primary flow is unaffected by the flow of coolant.

With the use of these assumptions and blade mean-section data, an integration was first made of the gain in kinetic energy output resulting from the expansion of the coolant from static conditions on the blade surface to the blade exit, the static conditions on the blade surface being obtained from the theoretical blade surface pressure distribution presented in reference 5. Next, an integration was made of the loss in kinetic energy output resulting from mixing the primary flow with the lower velocity coolant flow. The integrated effects of the coolant expanding and mixing with the primary flow were then summed to obtain the net output of the coolant flow.

RESULTS AND DISCUSSION

The investigation of the effects on stator performance of transpiration discharge of coolant through a self-supporting porous blade wall was made for nominal coolant flow rates of 0, 3, 5, and 7 percent of the primary flow. Mass flow and outlet flow angle results were obtained for a range of stator-inlet to downstream pressure ratios, while stator output and efficiency results were obtained at a constant stator-inlet to downstream hub pressure ratio of 1.72, which corresponded to stator design conditions (ref. 5).

To determine the effect of coolant discharge on the performance of a stator requires that the base noncooled stator performance be established. This base performance should exclude both the effect of coolant ejection and the effect of any blade modifications required for coolant flow.

For the subject blading, apertures in the blade surface are required for coolant flow. With zero-coolant flow, these holes, if left open, would permit some of the primary air to blow through the blading and would present surface discontinuities in the primary flow path that are not present in noncooled blading. Therefore, in the subject investigation, when establishing the performance of the noncooled stator on which the effect of transpiration coolant discharge is based, the coolant holes in the outer blade wall were sealed to prevent primary flow through the walls, and the surface was then smoothed flush to the outer blade wall to eliminate the surface discontinuities of the coolant holes.

The minimum transpiration cooling rate presented in the results is approximately 3 percent of the primary flow. This represents a minimum practical value of coolant rate for this blading and occurs when the coolant pressure inside the blading is slightly higher ($\sim$1 in. of Hg or 3.38×10^3 N/m^2) than the highest pressure on the blade surface (inlet stagnation pressure). Values of coolant pressure lower than stator primary air inlet stagnation pressure with corresponding lower coolant rates represent unrealistic operating conditions since some of the primary air flows abnormally through the interior of the blading.

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The results are presented in three sections. The first section presents the experimental results for the subject stator. The second section compares the experimental results for the subject stator with similar results for the stator of reference 3 having trailing-edge coolant ejection. The last section compares the experimentally determined results for the subject stator with those predicted analytically.

## Experimentally Determined Effects of Transpiration Coolant

Discharge on Stator Performance

The first part of this section concerns mass flow characteristics; the second part concerns pressure loss and stator output; and the third part relates to stator efficiency.

Effect of transpiration coolant discharge on stator mass flow characteristics. - The experimental mass flow of the stator with transpiration coolant flow is presented in figure 7. In figure 7(a), the equivalent primary air mass flow is shown as a function of the ratio of inlet total to average downstream pressure $p_0'/p_{d,ave}$ and coolant flow. The dashed curves at zero-percent coolant flow with the blade coolant holes open and at 3.5-, 5.0-, and 7.0-percent coolant flow were obtained from the crossplots of experimental weight flow data. The data for zero coolant with the blade coolant holes closed were obtained indirectly as described under TEST AND CALCULATION PROCEDURES. The solid curve shown was extrapolated from a single test point. From the figure, a trend of decreasing primary flow with increasing coolant flow is evident. The flow for zero coolant with the coolant holes open is also a little less than the flow with the coolant holes sealed.

In figure 7(b), the same flow data as shown in figure 7(a) are presented as a function of the average throat pressure ratio $p_0'/p_{1,ave}$. As in figure 7(a), a trend of decreasing primary flow with increasing coolant flow is evident. Examination of the data of figure 7(b) shows that, at a constant throat pressure ratio, the decrease in primary flow with coolant addition, relative to the flow with the coolant holes sealed, is a little larger than the increase in coolant flow. There is, therefore, a decrease in total flow with the flow of coolant.

In figure 8, the fractional decrease in total flow relative to the flow with the coolant holes sealed is shown as a function of coolant flow rate. The data shown are for a constant throat pressure ratio near stator design conditions. These results show the total flow to decrease about 1.5 percent between 0- and about 3-percent coolant flow. From about 3- to 7-percent coolant, which is the practical range of coolant flows for this blading, the percentage decrease in total flow remains nearly constant at 1.5 percent.

With transpiration discharge, a decrease in total flow with coolant addition is not

Figure 7. – Mass flow of stator with transpiration coolant.
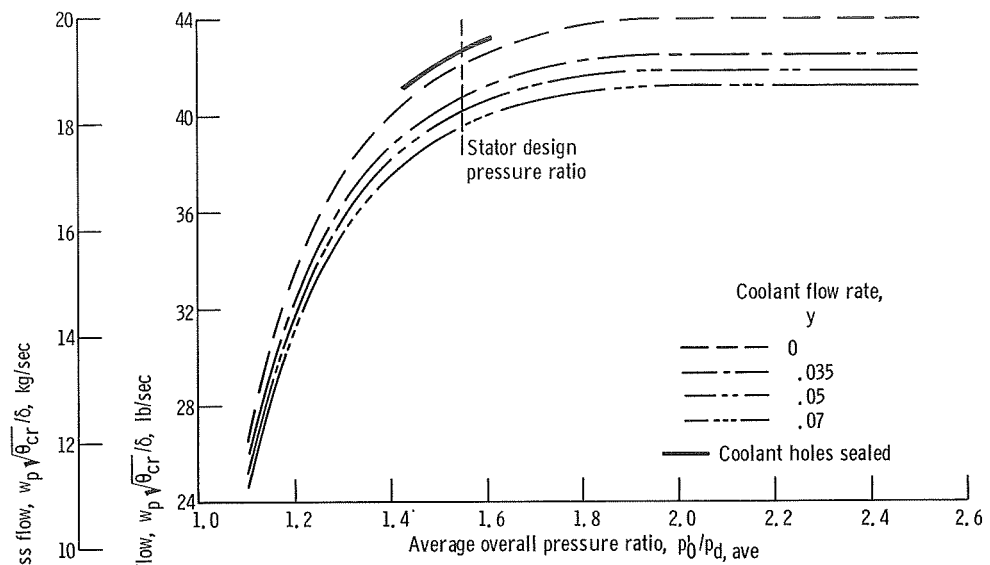
(a) Based on average overall pressure ratio.
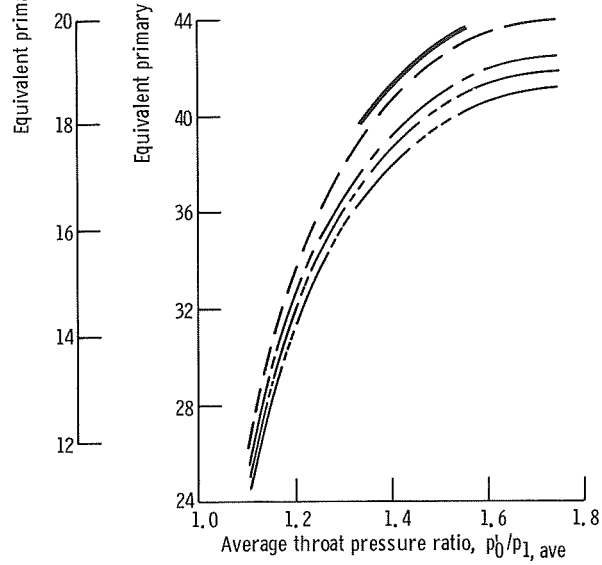
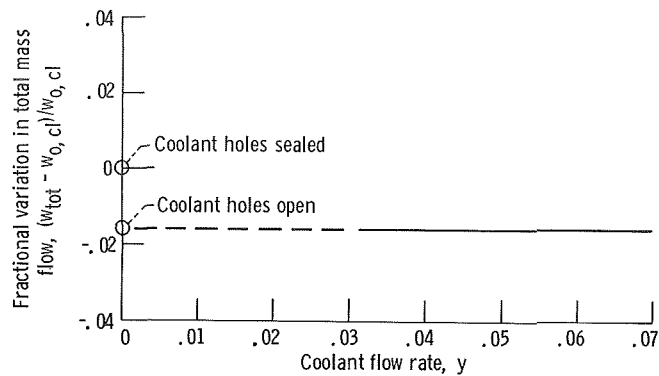(b) Based on average throat pressure ratio.

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Figure 8. - Fractional variation in total mass flow as function of transpiration coolant flow rate.

unexpected. Most of the coolant flow is added upstream of the throat, so the throat flow is about equal to the total of the primary and coolant flows. If the average throat velocity of the coolant were the same as that of the primary flow without coolant, the total flow would be constant for equivalent primary flow conditions and throat pressure ratio. However, at the throat, the average velocity of the coolant must be less than that of the primary flow. The reason is that the total pressure of the coolant in the direction of primary flow at the point of discharge on the blade surface is always less than the inlet total pressure of the primary flow. Therefore, the total flow is decreased with coolant addition.

In figure 8, a reduction in flow of about 1.5 percent, with zero-coolant flow and the holes open, is shown relative to the flow with coolant holes sealed. This reduction in flow was expected since, with the holes open, part of the flow is expanding inefficiently through the porous blading, whereas, with the holes closed, all the flow is expanding through the design flow path of the blading.

The stator outlet flow angles α obtained at the design outlet hub critical velocity ratio $(V/V_{cr})_{h,3}$ with various coolant fractions are shown in figure 9. (The data points
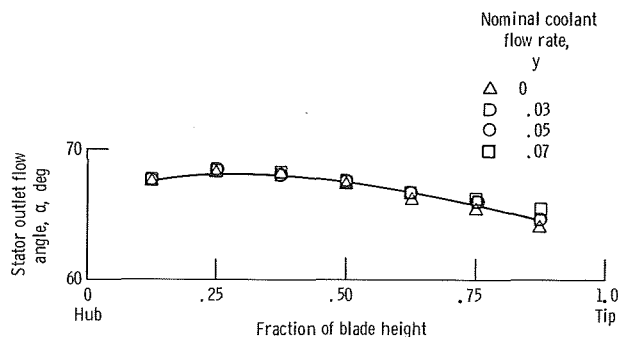

Figure 9. - Effect of coolant flow rate on stator free-stream outlet flow angle at design outlet hub critical velocity ratio of 0.896.

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shown were obtained from curves faired through average experimental free-stream angle traces.) The outlet flow angles are little, if at all, affected by coolant addition. Since, from continuity considerations, very small changes in flow angle will accommodate small changes in total flow, these results are consistent with those discussed in the last paragraph which showed the total flow to vary only about 1.5 percent with coolant addition.

Effect of transpiration coolant discharge on pressure loss patterns at blade exit and on kinetic energy loss. - The effect of transpiration discharge of coolant on wake pressure loss traces is shown in figure 10. The results shown are for a constant stator-inlet
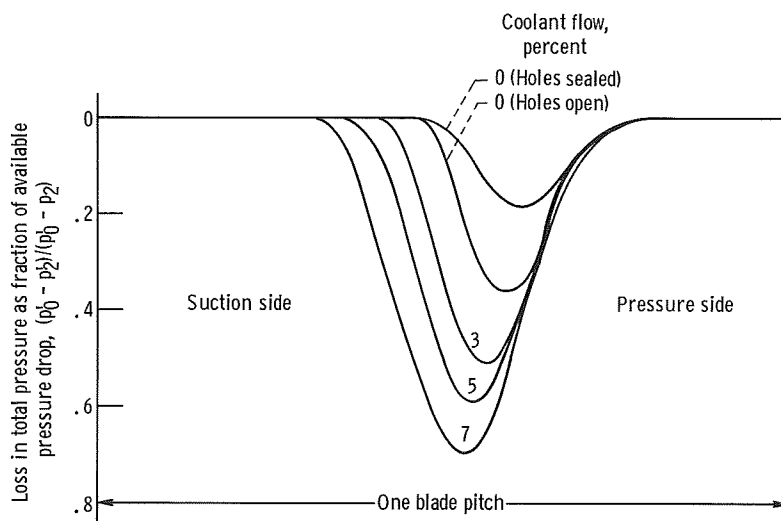
Figure 10. - Comparison of pressure loss data with different coolant flow rates. Data measured at blade mean radius.

to downstream hub pressure ratio of 1.72 corresponding to stator design conditions (ref. 5). (The traces presented are from circumferential traverses measured at the blade mean section to exclude the influence of end wall losses.) In this figure, the traces for 3-, 5-, and 7-percent nominal coolant flow are superimposed on the trace for 0-percent coolant flow to show the manner in which the wake patterns changed with coolant flow. The size of the wake area indicates the loss in specific kinetic energy of the total flow $\Delta h_{tot} = (w_p \Delta h_p + w_c \Delta h_c)/(w_p + w_c)$. That is, as the wake area increases, the specific kinetic energy output of the total flow decreases. The wake area with zero coolant and the coolant holes open is larger than that with the holes closed. Therefore, relative to the loss with the holes sealed, an additional loss is incurred with the holes open as a result of some of the primary air leaking through the interior of the blading and/or the surface discontinuities of the coolant apertures.

With increasing coolant flow, the wake area (fig. 10) also continually increases, so

the specific energy of the total flow decreases with increased coolant flow. The figure also shows that the increase in wake area occurs on the suction side of the blading. Possible implications of this will be discussed later in this section.

The decrease in specific output of the total flow is shown quantitatively in figure 11. These results, which are based on annular data, show the amount of decrease to be a little greater than the coolant fraction and, therefore, quite large at the larger coolant flows. For instance, at 5-percent coolant flow, the decrease in specific output of the total flow is 5.8 percent, and at 7-percent coolant flow, the decrease is 7.7 percent.

A better understanding of the effect of transpiration coolant ejection on stator output can be obtained from figure 12. In this figure, the fractional variation in specific output based on primary flow, instead of on total flow, is presented as a function of coolant flow rate. Relative to the results with the coolant holes sealed, the specific output per unit of primary flow $\Delta h_{pr} = (w_p \Delta h_p + w_c \Delta h_c)/w_p$ decreases about 1.0 percent at 3-percent coolant flow and then remains constant with increased coolant flow. This decrease is most significant since it indicates that any increase in output resulting from the flow of
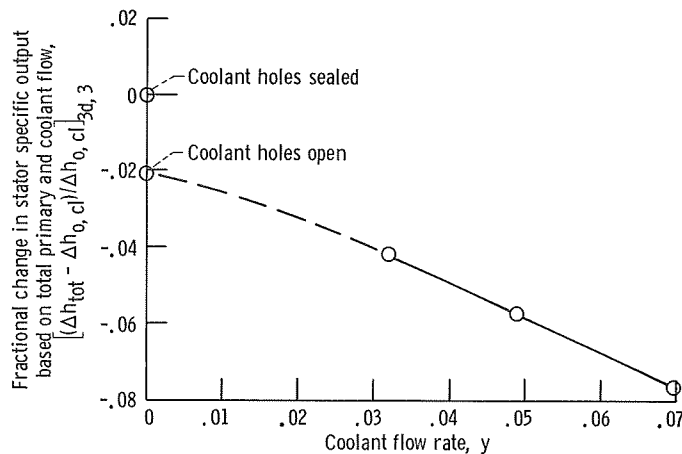

Figure 11. - Variation in stator specific output, based on total flow, with coolant flow rate.
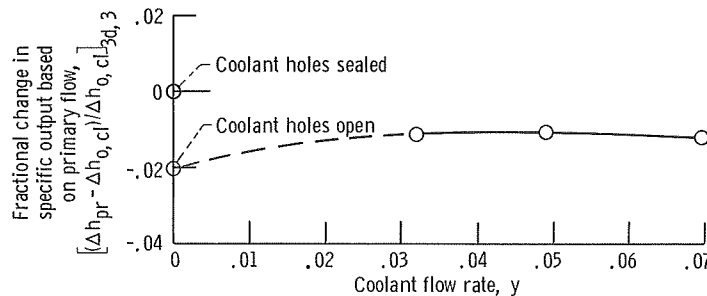

Figure 12. - Variation in stator specific output, based on primary flow, with coolant flow rate.

coolant is more than offset by a decrease in output due to the coolant flow mixing and interferring with the primary flow. Possible reasons for this occurrence are discussed in the following two paragraphs.

With transpiration cooling, the direction of the coolant at the point of discharge is generally perpendicular to the direction of the main gas stream. Therefore, at the point of injection, the coolant flow, having a nominally zero component of velocity in the direction of the main gas stream, contributes little, if any, useful energy to the flow. After being injected into the main gas stream, the coolant possesses ideal available energy corresponding to the static head between the point of discharge and the blade exit. However, on the suction surface of the blading, in the area of diffusion, this static head is not positive, but negative, since the static pressure at the point of discharge is less than the static pressure at the blade exit. As shown in reference 5, figure 13, for blading with this geometry, the diffusion area represents a considerable portion of the total blade surface. To discharge flow perpendicular to the main gas stream, at low velocity, in a negative pressure gradient would seem to contribute to a thickening (and possibly separation) of the boundary layer of the main gas stream. Some evidence that the flow of coolant causes such a loss was shown in figure 10, which indicated that the increase in wake area with increased coolant flow occurred on the suction (diffusion) side of the blading. It is also possible that the ejection of the coolant increases the friction loss of the primary flow, since the coolant jets may act as surface imperfections.

On the portion of the blade surface having positive static head between the point of coolant discharge and blade exit, the static head of the coolant is converted, at some efficiency, into kinetic energy that contributes to the useful output of the blade row. However, as discussed previously in the section Effect of transpiration coolant discharge on stator mass flow characteristics, the velocity of all portions of the coolant after expansion is less than that of the primary flow. As a result, some of the useful output of the blade row is lost because of the coolant flow mixing with the primary flow.

As shown in figure 12, the net result of the effects discussed in the preceding two paragraphs is that any increase in output resulting from the expansion of the coolant is more than offset by a reduction in output caused by the coolant flow mixing and interfering with the main gas stream.

Effect of transpiration coolant discharge on stator efficiency. - These results are presented in terms of both a primary air efficiency η_p and a thermodynamic efficiency η_t. The primary air efficiency relates the actual kinetic energy of the primary and coolant flows to the ideal kinetic energy of the primary flow only, while the thermodynamic efficiency relates the same actual kinetic energy of the combined flows to the sum of both the primary and coolant flow ideal energies. Values of thermodynamic efficiency presented are based on charging the coolant flow with the ideal energy between the coolant supply annulus and stator exit static pressure.
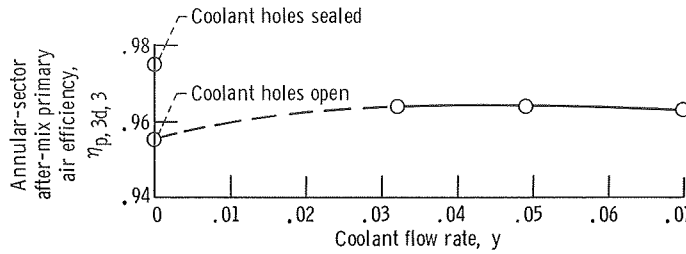
Figure 13. - Variation in stator primary air efficiency with coolant flow rate.

The variation of annular-sector after-mix primary air efficiency with transpiration coolant flow rate is presented in figure 13. Relative to the efficiency with the coolant holes sealed, the figure shows that the primary air efficiency decreases about 1 percent at about 3-percent coolant and then remains about constant between coolant flow rates of 3 and 7 percent.

The decreasing trend of primary air efficiency with coolant addition shown in figure 13 can be explained with reference to figure 12. The primary air efficiency η_p is equal to the specific output based on primary flow Δh_{pr} (see fig. 12) divided by the ideal specific energy of the primary flow $\Delta h_{i,p}$. For this investigation, with a constant primary air pressure ratio, the ideal specific energy of the primary flow is constant. Therefore, the primary air efficiency varies with coolant addition, just as the specific output based on primary flow varies for the same reasons.

In figure 14, experimentally determined values of annular-sector after-mix thermo-
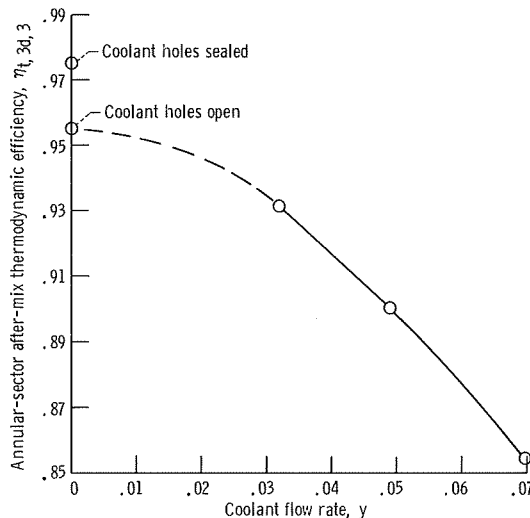

Figure 14. - Variation in stator thermodynamic efficiency with coolant flow rate.

dynamic efficiency $\eta_{t,3d,3}$ are shown as a function of coolant rate. The thermodynamic efficiency decreases sharply with coolant flow rate. For instance, with the coolant holes sealed the efficiency is 97.5 percent. At $3\frac{1}{4}$-percent coolant flow, the efficiency is about 93 percent, and at 7-percent coolant flow, the efficiency is about 85.5 percent, which represents decreases of about 4.5 and 12.5 percent, respectively, relative to the efficiency with the holes sealed. The much larger decrease in thermodynamic efficiency (fig. 14) relative to the decrease in primary air efficiency (fig. 13), of course, results from the fact that the thermodynamic efficiency is charged with the coolant flow ideal energy.

Comparison of Effects on Stator Performance of Transpiration Coolant

Discharge and Trailing-Edge Coolant Discharge

The first part of this section compares the effects of the two discharge methods on stator mass flow, and the last part compares the effects on stator overall efficiency. The trailing-edge coolant discharge data were previously obtained and reported in references 2 and 3. All data presented are near design stator pressure ratio.

Comparison of effects of transpiration and trailing-edge discharge on stator mass flow. - Figure 15 presents the fractional variation in total and primary flow for the two
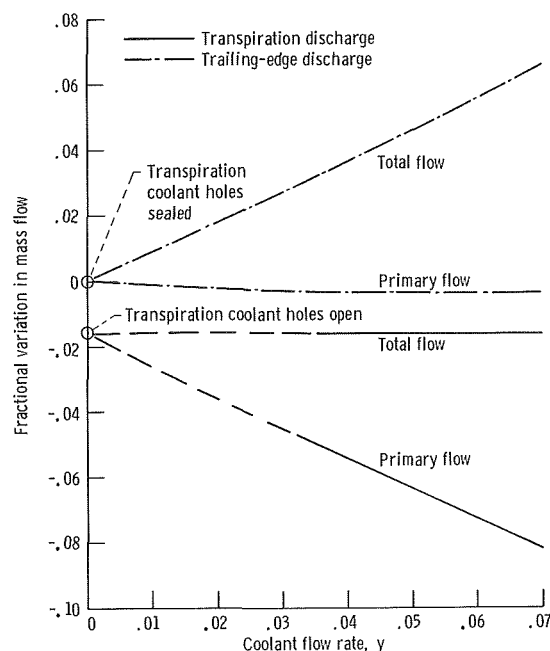

Figure 15. - Comparison of variation in mass flow as function of coolant flow rate for transpiration and trailing-edge coolant discharge.

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discharge methods as a function of coolant rate. As previously discussed with transpiration ejection, the primary flow decreases at a rate that is a little larger than the coolant flow rate, so the total flow is decreased a small amount with the addition of coolant.

With trailing-edge discharge of coolant, the variations in total and primary mass flow as a function of coolant fraction are quite different from those with transpiration discharge. With trailing-edge ejection of coolant, the primary flow is decreased only slightly, so the total flow increases significantly with increased coolant addition.

The principal reason for the different trends of the two methods of discharge is the different discharge locations. With transpiration discharge of coolant, most of the coolant is discharged upstream of the stator throat. The coolant then fills part of the available throat area so the primary flow is reduced. With trailing-edge ejection, the coolant is discharged downstream of the throat so the throat area does not have to accommodate the coolant flow and the primary flow remains nearly constant.

Comparison of effect of transpiration coolant discharge and trailing-edge coolant discharge on stator efficiency. - In figure 16, a comparison is presented of the fractional change in primary air efficiency as a function of coolant flow for both transpiration discharge and trailing-edge discharge of coolant. As with mass flow characteristics, the results are quite different for the two types of coolant discharge. With trailing-edge discharge, the primary air efficiency increases with increasing coolant flow; with trans-
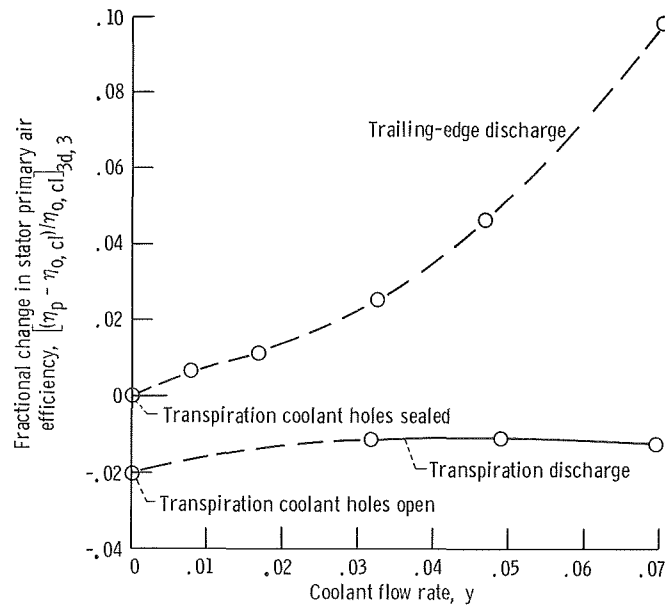

Figure 16. - Comparison of variation in primary air efficiency as function of coolant flow rate for transpiration coolant discharge and trailing-edge discharge.

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piration coolant discharge, the primary air efficiency is decreased because of coolant flow.

The different trends are caused primarily by the differences in the location and direction of ejection of the coolant for the two types of discharge. With transpiration coolant discharge, the coolant is ejected generally perpendicular to the main flow and so contributes little, if any, useful kinetic energy at the point of discharge. An increase in output can result because of the expansion of the coolant flow from the point of discharge. However, whatever portion of this output is realized, it is more than offset by a reduction in output resulting from the coolant flow entering and mixing with the main gas stream. The result is a net decrease in specific output per unit of primary flow. As discussed previously in the section Effect of transpiration coolant discharge on stator efficiency, for this investigation a given decrease in specific output per unit of primary flow is equivalent to the same decrease in primary air efficiency.

With trailing-edge ejection, the coolant is discharged at the downstream stagnation point of the blading and in the direction of primary flow. Since it is discharged at this location, the coolant flow has no apparent effect on the flow of the primary fluid, and, since it is discharged in the direction of flow, it contributes its kinetic energy to the useful output of the primary fluid. In addition, as explained in reference 3, there is a reduction in trailing-edge loss with trailing-edge ejection, which increases the specific output. For these reasons, with trailing-edge ejection, the specific output based on primary flow and, consequently, the primary air efficiency, continually increases as the coolant rate increases.

Summarizing, the results of this section indicate that, with transpiration discharge, the coolant flow contributes little, if any, useful output to the stator blade row and has an adverse effect on the specific output of the primary flow. With trailing-edge discharge, the coolant flow contributes to the useful output of the stator and apparently has no harmful effect on the specific output of the primary flow.

In figure 17, the effect on stator thermodynamic efficiency of transpiration and trailing-edge coolant discharge is compared. With transpiration discharge, the thermodynamic efficiency decreases rapidly with increased coolant flow, while with trailing-edge ejection, the thermodynamic efficiency first increases and then decreases.

Comparing the trend of thermodynamic efficiency of figure 17 with that of primary air efficiency in figure 16 for either of the two discharge methods reveals that the trend of thermodynamic efficiency falls below the trend of primary air efficiency by an increasing amount with increased coolant flow. This occurs, of course, because the thermodynamic efficiency is charged with increasing amounts of coolant flow ideal energy as the coolant flow is increased, and the primary air efficiency is not. Except for this fact, the reasons for the variations in trends in thermodynamic efficiency are the same as the reasons for the variations in trends in primary air efficiency.
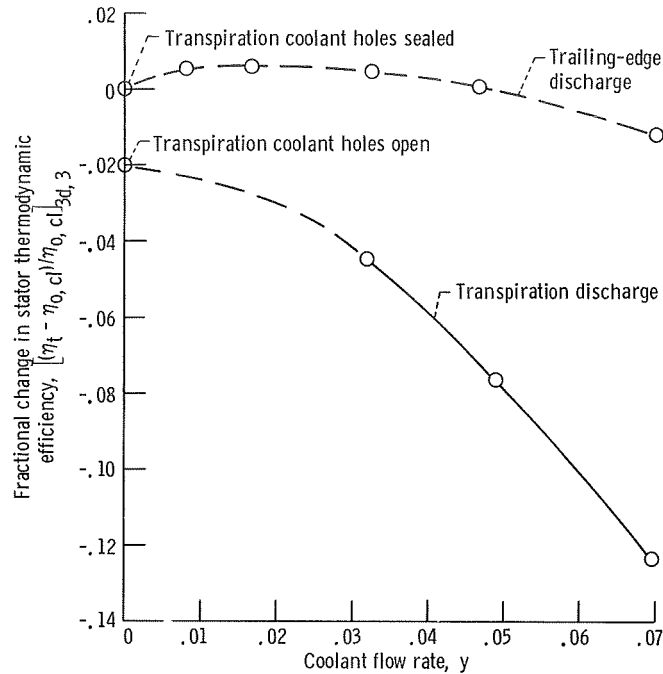
Figure 17. - Comparison of variation in thermodynamic efficiency as function of coolant flow rate for transpiration coolant discharge and trailing-edge discharge.

Comparison of Experimental and Predicted Effects of Transpiration

Coolant Ejection on Stator Efficiency

In a preceding section, TEST AND CALCULATION PROCEDURES, an analytical method for predicting the effect on stator efficiency of transpiration coolant discharge was described. In figure 18, the fractional variation in primary air efficiency for the
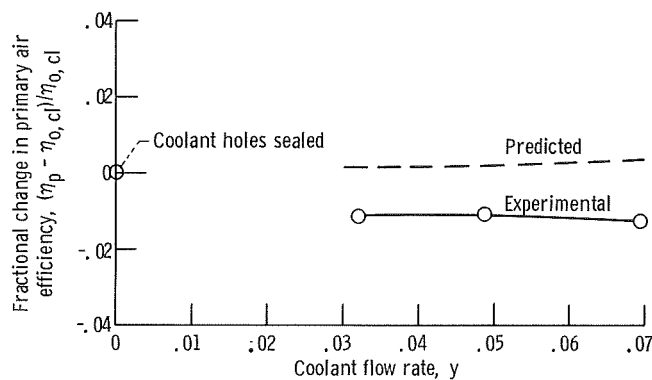

Figure 18. - Comparison of experimental and predicted changes in primary air efficiency for transpiration coolant discharge.

subject stator, predicted by using the described analytical method, is compared with the experimentally determined variation. In the range of coolant flows of interest, between 3 and 7 percent, the predicted results show about a 1/4-percent gain in primary air efficiency, whereas the experimental results show about a 1-percent loss in primary air efficiency. The predicted results indicate that the coolant flow contributes nearly zero output to the blade row, which is in agreement with experimental results. The dis-crepancy of around $1\frac{1}{4}$ percent apparently results from the fact that the prediction method assumes that the flow of coolant has no effect on the efficiency of the primary flow, while the experimental results show that the coolant flow adversely affects the efficiency of the primary flow.

SUMMARY OF RESULTS

An investigation was made to determine the effects on turbine stator performance of transpiration coolant ejection through a self-supporting, variable-porosity blade wall. The effects of transpiration discharge of coolant on stator mass flow, outlet flow angle, output, and efficiency were obtained for nominal coolant flow rates of 0, 3, 5, and 7 percent of primary mass flow. The effects on stator mass flow characteristics and outlet flow angles were investigated for a range of stator-inlet to downstream pressure ratios, while the effects on stator output and efficiency were investigated at a constant stator-inlet to hub downstream pressure ratio of 1.72.

Efficiency results are reported in terms of both primary air efficiency and thermo-dynamic efficiency. The primary air efficiency relates the useful kinetic energy output of the stator to the ideal energy of the primary flow, while the thermodynamic efficiency charges the stator with the ideal energies of both flows.

For this blading, operation between 0- and about 3-percent coolant flow is unrealistic, since, under these conditions, some of the primary air flows abnormally to the inside of the blading through the coolant holes. Therefore, to obtain realistic operation of the blading with zero-coolant flow, the results for zero-coolant flow were obtained with the coolant holes sealed. The findings for the subject stator are as follows:

1. With coolant flow rates between about 3 and 7 percent, the total mass flow was decreased a constant small amount of about 1.5 percent relative to the mass flow with the coolant holes sealed. The reasons for this decrease are that (a) most of the coolant is added upstream of the throat, so the throat area must accommodate most of the coolant flow, and (b) at the throat, the coolant velocity is less than the average velocity of the primary flow.

2. The stator-outlet flow angles were affected little, if any, by coolant flow rate, as expected from continuity, since the total mass flow was little affected by coolant flow rate.

3. Relative to the specific output with the coolant holes sealed, the specific kinetic energy output based on the primary flow was decreased a constant small amount of about 1 percent with coolant additions between about 3 and 7 percent. This result indicates that any increase in output attributable to the expansion of the coolant flow is more than offset by a decrease in output due to the coolant flow mixing and interferring with the primary flow.

4. Relative to the efficiency with the coolant holes sealed, for coolant additions between about 3 and 7 percent, the primary air efficiency decreased a constant small amount with increased coolant flow. And the thermodynamic efficiency decreased an increased amount with increased coolant flow. For instance, the decrease in primary air efficiency was a constant value of about 1 percent, whereas the decrease in thermodynamic efficiency at $3\frac{1}{4}$ and 7 percent coolant was about 4.5 and 12.5 percent, respectively. These decreasing trends in efficiency are directly related to the decreasing trend in specific output.

5. A comparison of predicted and experimental effects of transpiration discharge on primary air efficiency for coolant additions between about 3 and 7 percent showed the following: Relative to the efficiency with the coolant holes sealed, the predicted results indicated about a 1/4-percent gain in primary air efficiency, whereas the experimental results showed about a 1-percent loss in primary air efficiency. The discrepancy apparently is caused by the fact that the prediction method assumes that the coolant flow does not affect the primary flow efficiency, while experimental results show that the coolant adversely affects the efficiency of the primary flow.

6. A comparison of the effects on stator performance of transpiration and trailing-edge coolant discharge showed that, for the blading investigated, transpiration discharge caused a significantly larger loss in performance than trailing-edge discharge. The apparent reasons for the different effects of the two methods are the different locations and direction of coolant discharge of the two methods. With transpiration discharge, the coolant flow, discharged generally perpendicular to the blade row, contributes little, if any, useful kinetic energy output to the blade row and has an adverse effect on the specific output of the primary flow. With trailing-edge discharge, the coolant flow is discharged in the direction of flow which contributes to the useful output of the blade row with no apparent effect on the primary flow. In addition, evidence shows that the coolant flow reduces the blade trailing-edge loss.

Lewis Research Center,
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Cleveland, Ohio, June 1, 1970,
720-03.

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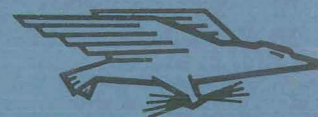
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